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CEMENT AND LIME MANUFACTURE

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Friction Loss in Cement Manufacture.

REVERTING to the article on "Fuel Consumption of the Rotary Kiln" and the diagram of temperatures given on page 32 of the March, 1942, number of this journal, it may have been noted that temperature difference and temperature drop, with resultant heat loss, occur at every stage of the process—from slurry to clinker—and, although it is not possible to avoid or prevent the whole of this loss, the amount could probably be reduced considerably by careful design and intelligent control. The same general principle is also applicable to the power consumption side, and detailed consideration will now be given to friction and friction losses.

One instance is on record where the overall consumption of a fairly large works amounted to between 160 to 170 kWh. per ton of cement made; this cement was of normal quality and fineness and was manufactured from a limestone of medium hardness and purity; the plant was a "straight" plant without any auxiliaries for dealing with or cleaning the raw materials. This power figure may be compared with a corresponding figure of 100 to 105 kWh. per ton for another works dealing with a similar material, or with, say, 80 kWh. per ton as normally obtained at a works using softer raw material and producing the same class of cement. It is possible that the apparent excess power used in the first-mentioned works was absorbed by friction, and this article may indicate the possible cause of this waste.

The friction that occurs in machinery is the cause of considerable loss, and nearly all the wear in the working parts may be attributed to it. The effect of friction is so well understood that the sponsors of ball and roller bearings are able to state a figure for the higher running efficiency of machines when fitted with these bearings; it can also be stated with a measure of truth that the friction loss of bearings of the ball or roller type will normally be only from one-fifth to one-tenth of the loss of bearings of the more usual kind. These figures, however, are to be regarded as only typical.

Friction is the cause of serious loss in the cement industry owing to the considerable amount of power used in the process and owing also, and perhaps more so, to the abrasive nature of the material at all or most stages of manufacture and to the difficulty of keeping the abrasive material out of the bearings. The conditions are not quite so bad with softer raw materials, but where the raw material is of the limestone class almost as much power is required, per ton ground, for preparing the stone as is required for dealing with the clinker.

We will take some of the principal units used in the manufacturing processes and consider how much of the power taken is used usefully and how much of it is wasted, and at the same time we may consider what can be done to reduce the latter. We must bear in mind that 1 b.h.p. for 7,000 hours a year is approximately equal to 6,000 kWh.; that this amount of power at 0·4 pence per unit has a value of about £10; and that 3 h.p., or 2·5 kWh., per ton of cement made is equivalent to 1d. per ton on or off the manufacturing cost.

Crushers.

Large and small crushers of any class will each call for up to 2 h.p. per ton for all the stone that passes through them; the figure, however, will be influenced by the size and classification of the feed as well as of the final product. It is possible that the hammer crusher under large reduction-ratio conditions, and for the production of a small size product, is the most efficient of all crushers and operates with the lowest friction losses; it is likely that the higher shaft speed and the lower bearing pressure at which this machine operates contribute in no small measure to this result. Friction is always lower at high speeds and higher at low speeds, especially when such speeds are used in conjunction with heavy pressures such as are common in the jaw, roll, and gyratory type crushers; there is no bearing pressure in the hammer machine comparable with the pressures usual in these other machines. Some makes of crusher are now fitted with ball or roller bearings, but experience in this matter is very limited at the moment or in any case the results have not been released for public information. But machines fitted in this way should run with very low friction losses with resulting low power consumption; they should also run for long periods without bearing trouble if the means provided for dust exclusion are really effective.

Papers read before the Institution of Mining and Metallurgy imply that the efficiency of all types of crushers is very low, but no indication is given of the best way of raising the efficiency. It can be confirmed, however, that if a small size product with a general fine classification is required, the hammer machine should always be adopted, assuming that there is no known condition that would prejudice its use. Special or selective classification of material of an intermediate size may call for machines with other characteristics in order to obtain the classification required, and in that case the friction loss or the h.p. per ton would be higher and have to be accepted.

Grinding Mills (Wet and Dry).

A grinding mill may be considered as a multiple crusher wherein each ball or element of media has a specific duty, i.e., that of crushing a particle of grit

of appropriate size. If the ball or media is too small or the piece of grit too large no crushing will be effected, though work will have been done and energy expended; if the ball is too large or the particle of grit too small, some energy will have been wasted although crushing has been effected. These instances of wasted energy occur possibly millions of times per hour in every grinding mill. It is impossible in mill operation to arrange for or ensure selective duty for each ball or media or for each size of ball or media, and for this reason energy will continuously be wasted and the efficiency of the process be low. Low efficiency of the grinding process is inherent in every known method or system adopted; the highest grinding efficiency will probably be obtained with tube-type mills when the size and classification of the media are determined by the method illustrated and described on page 60 of the May, 1942, number of this journal. Other causes of loss of power in grinding mills are friction of the trunnion or similar bearings and of the toothed driving gear. The friction of pinion-shaft bearings, where such shaft is used, is small, and for our present purpose will not be taken into account. The standard of design, construction, and lubrication of trunnion bearings and of toothed driving gear has been considerably improved in recent years, and these may now be classed amongst the most efficient parts of such machinery; this in turn has called for, and actually obtains, better maintenance. The usual formula for bearing friction, etc., will apply and may be used with confidence for obtaining the frictional power.

The laws of solid friction may be restated as follows: (1) Friction is proportional to the load or pressure; (2) friction is independent of the running speed; (3) friction is independent of the area of surface in contact; and (4) friction depends upon the roughness or smoothness of the mating surfaces; it may also be influenced by the hardness or the softness of the mating surfaces.

With regard to (2), it may be stated that static friction is greater than the friction of motion, and that friction at very low speeds may approach the static friction condition. As an example we will take a tube-type mill absorbing 800 h.p., having shell dimensions of 7 ft. 6 in. diameter by 40 ft. long, rotating at 22·5 revolutions per minute, and loaded with 55 tons of media. Other data given in the following may be assumed to apply and the final figure obtained for the frictional power of the trunnion bearings and also the loss of power in the driving gear will probably be substantially correct:

	Tons.
Weight of mill, empty	105
Weight of media	55
Weight of grit	8
Total weight on the bearings	168

Say, 376,000 pounds.

The trunnion bearings are 2 ft. 6 in. diameter, and the rubbing speed of the journals is $2 \cdot 5 \times \frac{22}{7} \times 22 \cdot 5$ ft., say, 176·5 ft. per minute.

As the journals are of high-class finish and the bearings of the self-aligning type, lined with white metal, a value of 0·015 will be taken for the coefficient of friction, and the frictional power will be about

$$\frac{376{,}000 \times 176 \cdot 5 \times 0 \cdot 015}{33{,}000} = \text{say, } 30 \text{ h.p.}$$

On the basis of (1), if the total weight had been 25 per cent. greater, the frictional power of the trunnion bearings would increase by the same percentage. Also, if the trunnion journals had been 27 in. diameter instead of 30 in. diameter, the frictional power of the bearings, owing to the reduced length of rubbing surface per minute, would be 10 per cent. less than that given.

The friction of the spur driving gear will be influenced by the condition of the engaging and sliding surfaces of the teeth and by the class of lubricant used and the method of its application. It will also be influenced by the length of sliding engagement of the teeth, which in turn is determined by the depth of tooth, but depth of tooth is determined by the dimension for the circular pitch. It thus appears that the friction loss will increase or decrease with the length of the circular pitch, other conditions remaining the same. A gear having a fine circular pitch will, therefore, run at a higher efficiency than a similar gear having a coarse circular pitch, and the circular pitch of the gear teeth should thus be as small as the operating conditions will permit.

So far as is known, no efficiency tests have been carried out on coarse-pitch gears, but only on fine-pitch gears. Two 600 b.h.p. totally enclosed D.B.S. increasing or reducing gears, fitted throughout with ball or roller bearings, and having splash lubrication, tested at the makers' works, gave an overall efficiency of 98·9 per cent.; that is to say, the loss was rather more than 1 per cent. of the power transmitted. Each gear thus ran with a total friction loss of between 6 h.p. and 7 h.p. As a matter of interest, a brief description is given of a method of carrying out such a test. The gears, which are exact duplicates, are arranged back-to-back, as indicated in *Fig.* 1, one gear being run as a reducing unit, and the other as a speeding-up unit. Temporary shafts BC and DE are provided, and the high-speed shaft of the reducing gear is extended through both sides of the casing for the same purpose. The procedure for coupling-up and obtaining the required tooth pressure may be as follows: Referring to *Fig.* 1, couplings A, B, C and D are coupled up in the usual way, but face (a) of coupling (E) is left undrilled. Coupling A is then locked, and a "torque," equal to full-load torque, is applied to face (b) of coupling E, and maintained while the bolt holes in face (a) of this coupling are drilled and the bolts fitted. When this has been done, the torque may be removed from E and the initial lock from A. The gears are then free to rotate, although they would require an amount of effort owing to the friction of the teeth under pressure conditions. The torque, or power, now required to rotate the gear light, at speed, would be exactly equal to the friction of the gear under full load and speed conditions.

The formula or rule for connecting the horse-power with speed (r.p.m.) and torque is:—

$$\frac{\text{H.P.}}{\text{R.P.M.}} = \frac{\text{T}}{63{,}025}\ ;$$

$$\text{and T} = \frac{\text{H.P.} \times 63{,}025}{\text{R.P.M.}}$$

In the formula, T is expressed in lb.-in., and the figure 63,025 is a constant.

The total power exerted by the motor under test conditions would be equally divided between the two gear units, and indicates the friction loss. The motor need be only a small one, and owing to this it should be possible to arrive at the power with exactness. In the D.B.S. test already mentioned, the circular pitch of these gears was possibly fully 0·7 in., and this small pitch resulted in only a small length of sliding contact between the faces of the teeth. A description

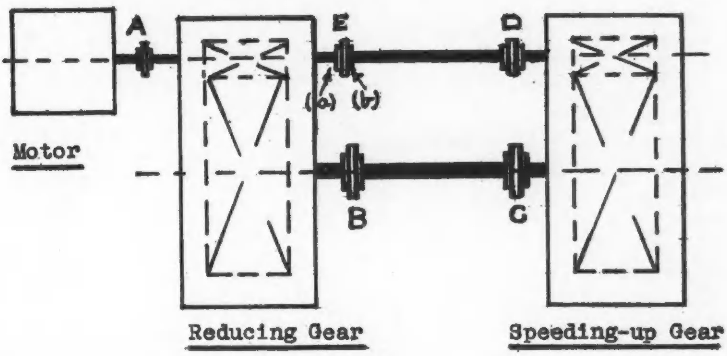

Fig. 1.

of these D.B.S. gears and the principal data of the test are given in *The Engineer* for October 20, 1933.

Two similar gears, but rather smaller, were tested at the Manchester works of another large manufacturer, and a still higher efficiency was obtained; the circular pitch of these latter gears, which were of the turbine type, was smaller than that of the D.B.S. gears.

As toothed gears having a circular pitch of fully 0·7 in. ran with a friction loss of about 1 per cent. under ideal conditions, it is possible that a loss up to 3 per cent. or 4 per cent. will take place with the usual pinion and girth gear of coarser pitch under the best conditions obtainable in cement mills. The friction loss, with the usual arrangement of mill drive and under best conditions, may thus be, say, 1 per cent. for the totally enclosed reduction gear, and 3½ per cent. for the pinion and girth gear; the total loss will thus be 4½ per cent. of the total power transmitted. If a trunnion drive were adopted, involving the use of a double- or triple-reduction high-class or precision gear, the friction loss would average, say, 1 per cent. per reduction, and the overall loss for the drive would

be, say 2 per cent. or 3 per cent., depending upon the number of reductions adopted.

Taking the pinion and girth gear with totally enclosed reduction unit as typical, the total frictional loss of the 800 h.p. mill will be as follows:

Totally enclosed reduction gear	1 per cent.	= 8 h.p.
Pinion and girth gear	$3\frac{1}{2}$,, ,,	= 28 h.p.
Trunnion bearings		= 30 h.p.
	Total friction loss ..	66 h.p.

A friction power of 66 h.p. enables 734 h.p., say, fully 90 per cent. of the total power, to be put into the useful performance of the mill, but the internal arrangement of the mill, the form of lining, and the classification of the media, jointly determine whether this power will be used to best advantage; the figures taken indicate a high mechanical efficiency.

Owing to the possibility of adopting finer pitched teeth under totally enclosed gear-box conditions, there is much to be said in favour of the central drive when site conditions permit of the extra length of space necessary and the higher capital cost is permissible. The difference in the friction loss of the two drives is as given in the following, the figures being based upon a mill absorbing 800 h.p. and working 7,000 hours a year; the single reduction gear in the first column and the third reduction in the second column are omitted, as they each have a value of 1 per cent.:

	Pinion and Girth Gear	Central Drive
Total mill power	800 h.p.	800 h.p.
Friction loss	$3\frac{1}{2}$ per cent.	2 per cent.
	= 28 h.p.	16 h.p.

Difference, 12 h.p. = 10 kW.

This difference in power for 7,000 hours a year with the cost of power at 0·4 pence per unit is equivalent to a total value of about £115 a year in favour of the central drive.

It need hardly be pointed out that it proves far easier to maintain a 100 per cent. grit exclusion condition with the central drive than under pinion and girth wheel conditions; shaft alignment and good lubrication conditions are also taken better care of.

Rotary Kiln.

The rotary kiln in its simplest form is a tube shell 6 ft. to 11 ft. in diameter, or even larger, and, say, 200 ft. to 500 ft. long, arranged at a slope of about one in twenty-five and rotating at about 1 r.p.m. The shell would be lined with firebrick, the slurry fed in at the upper end, drying and burning would take place inside, and the clinker discharged at the lower end. The kiln is supported on a number of riding rings which in turn rest on pairs of rollers with axles and bearings. The spacing of the rings may be from 50 ft. to 80 ft., or even 90 ft. The limiting pressure between the rings and the rollers is about six tons per inch of width, and the limiting pressure on the bearings about 500 lb. per square inch of projected area. The diameter of the rollers is 0·28 to 0·29 times the

diameter of the rings that rest upon them, and the diameter of the axle journals is 0·26 to 0·27 times the diameter of the rollers.

The principal point at which friction occurs during the kiln's rotation is at the axles and the bearings in which they run, but there are other though less important points, such as the thrust roller or similar equipment, and the inner surface of the riding rings that engage with (or rest upon) the kiln shell seatings that carry the load or weight.

The thrust of a kiln is a factor of the total revolving weight and of the slope of the kiln as it rests upon the supporting rollers. The thrust pressure of a kiln with a slope of 1 in 25 would be, say, 4 per cent. of the revolving weight; a figure for this is easily obtained by calculation; in practice, however, the total thrust is divided between the thrust roller proper and some or all of the load rollers, and the result is invariably a compromise. The part of the thrust transmitted to the load rollers is effected by "cutting" these rollers an infinitely small amount, but precision in the amount of cut can only be obtained when the surfaces of the riding rings and of the rollers are in good or perfect condition; moreover, even a fractional amount of cut may prove sufficient to disturb or entirely upset the running of a kiln, the result being that some of the rollers thrust the kiln upwards and some thrust it downwards, which results in undue friction and excessive wear. For the present purpose we will take the frictional power caused by the thrust to be 4 per cent. ($\frac{1}{25}$) of the power absorbed by the whole of the bearings carrying the load rollers.

With regard to the frictional rub or creep of the riding rings on their seatings, it will be understood that these rings cannot be made to fit as an amount of clearance (freedom) is necessary. To provide this freedom the ring is bored larger than the diameter of the seating, and the clearance that results sets up a condition of "creep" which results in continuous sliding, with friction. As wear takes place the clearance increases as also does the amount of creep with additional frictional loss. There appears to be no means of avoiding this condition and result, and as no figure for creep is available an arbitrary figure of 3 per cent. of the total load roller power will be assumed.

We will now take a set of imaginary dimensions and data for a kiln and, making reasonable assumptions, work out the possible power required and assign a figure for the frictional or lost power as calculated. The figures are not taken from practice though they may be near to those obtained under working conditions. The data and figures are as follows:

Average diameter of riding rings	16·0 ft.
,, ,, ,, rollers	4·6 ,,
,, ,, ,, axle journals	14·0 in.
=	1·2 ft.
Speed of kiln	1·0 r.p.m.
,, ,, rollers with axles (approx.)	3·5 ,,
Rubbing velocity of axle journals = (circumference × r.p.m.)	13·2 ft. per min.

Equivalent total pressure on bearings, 1,200 tons, say	2,688,000 lb.
Assumed co-efficient of friction	0·025
Bearing friction and h.p. at 1 r.p.m. = weight (lb.) × circumference (ft.) × co-efficient.	
$\frac{2{,}688{,}000 \times 13{\cdot}2 \times 0{\cdot}25}{33{,}000}$ = say	26·5 h.p.

The slow running speed and the heavy loading of the bearings indicate that a factor of 0·025 for the coefficient of friction would be more appropriate than the factor of 0·015 used previously for the mill trunnion bearings.

The drive for a kiln is usually obtained through the medium of a large girth gear—fixed to the kiln shell at about mid-length—and pinion, and also through two other pairs of gears. These gears would have an increased circular pitch of the teeth as observed from the driving end. The two first pairs of gears are usually bath-lubricated, but the large gear has its lubricant applied by hand intermittently. The possible efficiency of these three pairs of gears would be, say, 96 per cent., 95 per cent. and 90 per cent. respectively, and result in an overall gear efficiency of about 82 per cent. It need hardly be pointed out that applying lubricant to the third gear in the manner named is far from the best method, especially when searching for high efficiency.

The volume of the inside of the lining of the kiln might be, say, 16,000 cub. ft. and the amount of material, say, 9 per cent. of the internal volume. If the average weight of this material is taken at 90 lb. per cubic foot the total weight of the material would be 16,000 × 0·9 × 90, say, 130,000 lb. This material would be continuously lifted, say, 2 ft. or 2·5 ft. during the kiln's rotation, and the whole of the material would be lifted probably two or two-and-a-half times during each revolution of the kiln. The power required for this, taking the mean of the last two figures, would be, say, 130,000 × 2·25 × 2·25 ft. lb. per revolution, and the horse power at one revolution per minute would be

$$\frac{130{,}000 \times 2{\cdot}25 \times 2{\cdot}25}{33{,}000} = \text{say, } 20 \text{ h.p.}$$

A review of all the power figures for the kiln is as follows:

Lifting the material	20·0 h.p.
Bearing friction	26·5 ,,
Thrust roller friction, 4 per cent. of (2), say, ...	1·0 ,,
Creep friction, 3 per cent. of (2), say	1·0 ,,
Actual gear output	48·5 ,,
Gear input, 48·5/0·82	59·5 ,,
The gear loss is thus	11·0 ,,

Summarising the figures, we arrive at or deduce the following:

Bearing and thrust roller friction ..	27·5 h.p.	= 46·0 per cent.
Creep friction	1·0 ,,	= 2·0 ,, ,,
Gear friction	11·0 ,,	= 18·5 ,, ,,
Useful power	20·0 ,,	= 33·5 ,, ,,
Total friction loss	39·5 ,,	= 66·5 ,, ,,
Total power	59·5 ,,	

The low mechanical efficiency indicates the necessity for reducing or keeping down all the friction loss possible, especially that of the bearings and gear, as these alone account for up to or over 66 per cent. of the total power used. It must be borne in mind also that the basis figures used as the efficiency figures for both the gears and the bearings are possibly as high as practicable for machinery of this class; this implies that many kilns may be working at an efficiency much lower than that arrived at in the calculation, in which case the overall loss would be higher than that given. A case is on record where the changing of the driving gear of a kiln, and overhauling and realigning the roller bearings, resulted in the total power input being reduced from 110 to 115 h.p. to 60 to 65 h.p. as soon as the kiln had settled down to work under the new conditions; the difference in these figures with the cost of power at 0·4 pence per unit and the kiln working 7,000 hours a year represents, say, £450.

Other Plant.

Having dealt with crushers, wet and dry grinding mills, and the rotary kiln with cooler, it appears that the coal mill and the small auxiliaries alone now warrant consideration. The coal mill as a unit must be especially suitable for the characteristics of the coal generally used and of the usual moisture content. The fineness or the residue of the ground coal on the coarser sieves must be such that no sparks or stars are observable when the coal is blown into the combustion zone of the kiln, and the horse power per ton ground to obtain this condition should not exceed, say, 18 or 19 for the softer varieties and up to, say, 21 or 22 for the harder varieties; these figures, however, must be regarded as tentative or approximate.

The remaining units are small and they can possibly best be dealt with as a group rather than separately, and in principle rather than in detail. It must be borne in mind, however, that some or all of the small units may be the cause of considerable loss.

If the plant is already installed it may prove costly to raise the efficiency substantially, or it may not prove so easy as it would be in laying out a new plant; or, owing to the existence of a number of belts with shafting, it may be more economic to cut out the belts with shafting and substitute unit drives. The efficiency value of unit drives is easily obtainable, whereas it is much more difficult to ascertain exactly what the loss of a group of belts and shafts really is. It might be possible to isolate such a group and arrange a temporary drive for the purpose of obtaining the information; the work would need only to be done once and firm figures would be obtained. With care and reasonable expenditure it should be possible to reduce the auxiliary power of an old plant to about the same figure as that of a new installation; it might also provide the means for more convenient operation as well as a reduction of the running cost.

Worm reduction units are frequently adopted for unit drives, and a good idea of the running efficiency of such units may be obtained by a careful examination of *Fig.* 2. This curve indicates that the efficiency of these gear units falls off considerably at the higher ratios, and this must not be lost sight of in making

the selection; apart from this, there is much to be said in favour of this type of drive.

When large power stations were first being developed it seemed at first sight somewhat striking that industrial companies burning large quantities of fuel and using large quantities of heat should seriously consider the purchase of power for their own use, but the lower cost of power as now obtained from the power supply companies and the less amount of fuel used for the same duty fully confirm that, normally, the purchase of power, produced in bulk, is the correct procedure. The average fuel consumption of industrial companies is possibly up to, say, 3 lb. per electrical unit generated, whereas the public supply companies can produce electricity for an average consumption of between 1 lb. and

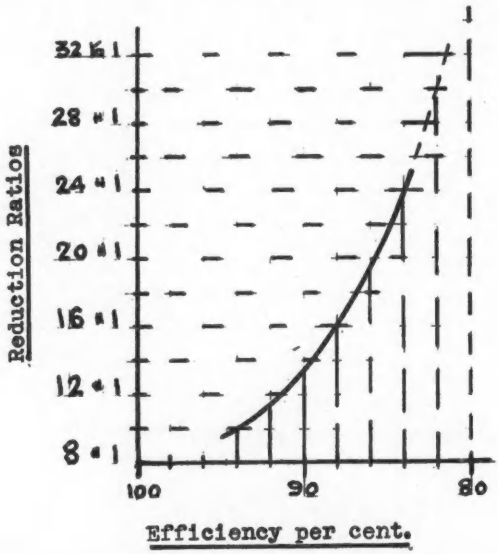

Fig. 2.—Probable Form of Efficiency Curve for Worm and Reduction Gear.

$1\frac{1}{2}$ lb. of coal per unit. The difference between these figures indicates a saving of up to, say, 1 cwt. of fuel per ton of cement produced.

With regard to the amount of power used by auxiliaries, a graph given by Sir Leonard Pearce in the twenty-sixth "Hawksley" Lecture before the Institution of Mechanical Engineers early in 1940 indicates that the average power used by auxiliaries in a group of the largest power stations in Great Britain may amount to 20 per cent. of the total power generated at those stations; this figure for auxiliary power appears very large, but must be considered above question. A consideration of the conditions that obtain in a modern cement plant suggests that this same ratio may apply there also. This auxiliary figure for power would cover all elevating and conveying and the intermediate transport of material,

also air and gas movement and pressure; water and slurry movement and pressure, and for the hundred-and-one uses that call for power at every point in the process for maintaining good conditions during stocking and for packing machinery. The figure at first sight appears large, but it has been checked for one or two modern plants and is found to be substantially correct; it will also be borne in mind that the amount of power used for auxiliaries is on the increase.

There is no objection to the use of a large amount of auxiliary power provided that it is used usefully, and provided also, and much more so, that it enables the process units, especially if they are large, to operate continuously and at high efficiency. If this latter proves to be the case, then a high consumption of auxiliary power is fully justified; in fact, anything that raises the efficiency ultimately saves power and fuel and should have careful consideration.

INDEX TO VOLUME XV, 1942.

THE index to Vol. XV, 1942, of this journal is printed at the end of this number, instead of separately as in the past. Due to the shortage of materials and labour, we regret that our binders are unable to accept orders for supplying cases or binding volumes of this journal for the time being.

Retarding the Setting of Cement.

An American patent has been granted to Messrs. E. Gruenwald, H. R. Durbin, and W. H. Tilley, of New York, for the use of casein in retarding the setting of cement. According to this patent, by adding from two-tenths to four-tenths of 1 per cent. of casein to concrete the setting rate is slowed down to any desired period. The patent has been assigned to the Lone Star Cement Corporation.

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The Use of Hydrated Lime for Thermal Insulation.

As the value of a material for thermal insulation is largely determined by the number of air-cells it contains, and as hydrated lime has this property in a high degree, some experiments on the possibility of using hydrated lime for thermal insulation have been made by Professor C. Mathers, of Indiana University, and Professor H. S. Wilson, of Louisville Municipal College. The problem was to prevent the setting of lime due to dampness, and this was achieved by adding to the lime about 0·5 per cent. of resin. The material was mixed in a ball-mill for two hours, and was then water repellent. Both high-calcium and dolomitic limes were used in the experiments, and the same results were obtained. Lime mixed with resin floated on water for several weeks, and could only be made to mix with water by heating to a high temperature or by vigorous stirring.

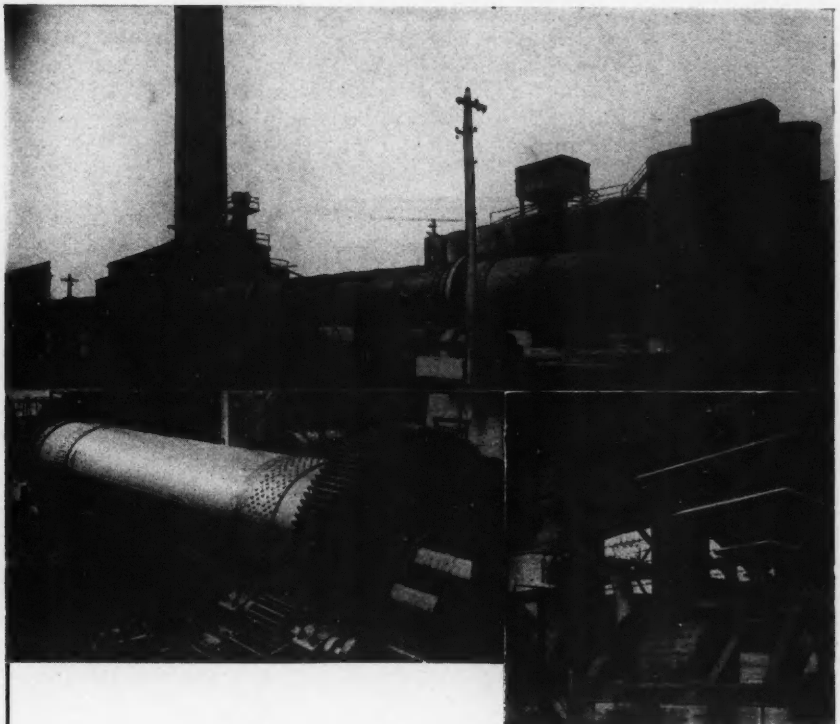
Tests were made to compare the insulating properties of the treated lime (which is called "insulim") with other materials used for thermal insulation. The method of testing was to place each insulating material in a beaker with a thermometer at a definite distance from the sides and bottom. The beakers were then placed on supports in an oil bath which was heated. A comparison of the rates of temperature rise at equal intervals of time gave the ratio of the insulating values of the various substances. The oil bath was more uniform than electric plates, ovens or refrigerators.

Each thermometer for the experiments detailed in *Table* 1 was 3·5 cm. from the bottom and 4·5 from the side of the beaker. The table shows that only "red mud," a by-product from alumina manufactured out of bauxite, gave as low a temperature rise as did the insulim, and all the other agents allowed more rapid rise.

TABLE I: TEMPERATURE AT VARIOUS PERIODS IN DEG. F.

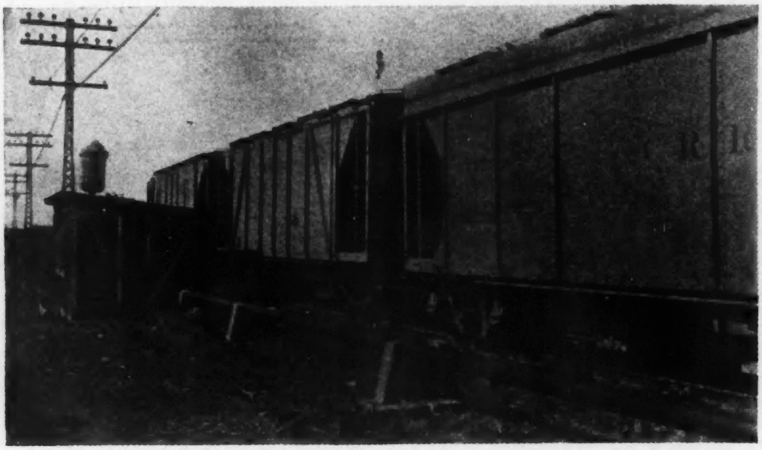
Time in minutes	Insulim	Sawdust	Rock wool	Red mud	Coarse cork
0	75	75	75	75	75
15	84	105	93	86	105
30	100	120	113	100	122
45	111	125	123	111	123

A long comparison test against a commercial insulating material, made by grinding waste gypsum board, showed marked superiority for insulim. The thermometer, in each case, was 3·5 cm. from the sides of the beaker and 4·5 from the bottom. It was estimated that the insulating values of insulim and gypsum scrap were in the ratio of 4·5 to 1. This was tested by having the thermometer bulb 1 cm. from the bottom of the beaker of insulim and 4·5 cm. from the bottom of the gypsum scrap. This showed that insulim is equal to about 4·5 times its thickness of the commercial gypsum waste as both thermometers remained approximately the same. A similar experiment, except that the thermometer in the gypsum waste was 2 cm. from the bottom, gave a slower rise for the thermometer in the insulim; this proves that 2 cm. of insulim is much better than 4·5 cm. of gypsum waste.



A direct comparison was made with rock wool in the same way. Again the thermometer bulb in the insulim was 1 cm. from the bottom of the insulim and 4·5 cm. in the rock wool. The result showed that 1 cm. of insulim is very much better than 4·5 cm. of rock wool. The rock wool was packed tightly in order to give it a maximum insulating value.

Bulk Delivery of Cement.



The above illustration shows the method of delivering cement in bulk to a central mixing plant in New York. The cement is delivered in trucks and pumped through a 600 ft. pipeline to a storage bin of 85 tons capacity. A 6 in. Fuller-Kinyon pump is used.

"The Concrete Year Book."

THE 1943 edition of "The Concrete Year Book" (edited by Oscar Faber, O.B.E., D.Sc., M.Inst.C.E., and H. L. Childe) is now ready (price 5*s.*; by post 5*s.* 8*d.*). The volume, which is now in its twentieth year, comprises 790 pages, and, as usual, includes a Handbook of information of everyday use to those engaged in concrete design and construction; a complete Directory of the concrete and allied trades and professions, and an extensive Catalogue illustrating and describing businesses, materials, and plant engaged in or used in the concrete and allied industries. The Handbook has been revised and brought up to date, and has new features, including new wartime uses of concrete and reinforcement tables. The Directory, revised up to December, 1942, gives temporary wartime addresses. Copies may be obtained from Concrete Publications, Ltd., 14, Dartmouth Street, Westminster, S.W.1.